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# RESEARCH MEMORANDUM

EFFECT OF BLADE-ROOT FIT AND LUBRICATION ON VIBRATION  
CHARACTERISTICS OF BALL-ROOT-TYPE  
AXIAL-FLOW-COMPRESSOR BLADES

By Morgan P. Hanson

Lewis Flight Propulsion Laboratory  
Cleveland, Ohio

**NATIONAL ADVISORY COMMITTEE  
FOR AERONAUTICS**

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EFFECT OF BLADE-ROOT FIT AND LUBRICATION ON VIBRATION

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SUMMARY

The vibration characteristics of several symmetrical ball-root-type blades simulating the mass and natural frequency of axial-flow-compressor blades were investigated under various mounting conditions by subjecting them to controlled periodic excitation and centrifugal loading. Looseness of blade mounting reduced vibration amplitude at moderate rotor speeds but lost its effectiveness as centrifugal force effectively tightened the blades at speeds comparable to normal turbojet-engine speeds. Use of solid lubricants such as molybdenum disulfide and graphite, however, extended this effect of looseness to speeds equivalent to that of normal engine operation. Although molybdenum disulfide oxidizes readily at temperatures above 900° F, it has satisfactory lubricating characteristics at the temperatures encountered in normal compressor operation. Lubrication also minimized the effects of fretting corrosion observed in the unlubricated loose-blade mounting. Blade-root configuration appeared to have an appreciable effect on damping characteristics of the blades.

INTRODUCTION

The conventional axial-flow-compressor blade, which is a relatively long cantilever with low bending stiffness in one direction, is susceptible to vibrations in several bending modes. Consequently, blades that are aerodynamically satisfactory are not always satisfactory with respect to vibration considerations. Blades of a new design sometimes vibrate excessively and frequently require extensive development to minimize the vibration difficulties. In some cases, the difficulties have been remedied by increasing the thickness of the blades. This type of correction is expensive, however, because of the increase in weight and the possible reduction in

compressor performance. Another method of eliminating excessive vibration is that of increasing the damping of the blade system. A blade with considerable freedom in the mount may produce damping due to friction between the sliding surfaces.

The suspected restriction of freedom in the mounting due to centrifugal loading in the rotors led to an investigation of blade-root damping at the NACA Lewis laboratory. As a simple means of simulating compressor-blade vibration without the complications of a complete engine, a simulated single-stage rotor was used in a spin rig in which it was possible to control induced vibration. Vibratory stresses encountered with varying exciting force at various rotor speeds were measured by resistance-wire strain gages on several ball-root-type symmetrical blades to compare the damping of blades that were tightly mounted, loosely mounted, or loosely mounted and lubricated.

#### APPARATUS

The three types of ball-root blade used in the investigation are shown in figure 1. The blades are rectangular in cross section from hub to tip and are so proportioned as to simulate the mass and the natural frequency of the blades of an intermediate stage of a currently used turbojet-engine compressor. The ball roots differ from each other in that two are of circular section but of different diameters, whereas the third has a double circular section. The large, small, and double ball-root sections have diameters of 0.389, 0.310, and 0.221 and 0.162 inch, respectively. The blades are made of 12-percent chrome steel, which is a commonly used compressor-blade material, with ground surfaces and a Brinell hardness of 225. The blades measure 1.9 inches from hub to tip and 0.602 inch across; the thickness tapers from 0.103 inch at the hub to 0.068 inch at the tip.

The rotor diameter was so chosen as to permit duplication of the maximum centrifugal force present in the reference compressor at the rated speed of the driving equipment used. The rotor, made of 14ST aluminum, is 13 inches in diameter and 0.602 inch thick. Slots in the rotor provided a mount for each blade type and conformed to the dimensions of the individual blade roots except for the case of tightly mounted blades, in which the fit was held to 0 to 0.0005 inch, whereas a clearance of 0.002 to 0.003 inch was maintained on all surfaces in the case of loosely mounted blades. With this clearance, it was possible to have a free tip movement of about 0.020 inch. The blades were secured in the axial direction

by means of a locking wire that engaged both the blade and the rotor. This method of retention, used with reasonable precaution, had an insignificant effect on the looseness of the blade mountings.

The rotor was driven by a variable-speed motor through a 3:1 speed increaser to a rated speed of 15,000 rpm. The assembly is shown in figure 2. A circular housing enclosing the rotor (not shown) reduced possible vibration excitation from external forces. Forced vibration excitation of the blades was produced by an air jet from a single 3/8-inch nozzle mounted on one side of the rotor at the tip radius and directed at the tip of the blades at an angle of 45° to the plane of rotation.

Each of the blades investigated was instrumented with two 1/8-inch, 120-ohm wire strain gages so mounted at the base of the blade as to measure maximum stress in the first bending mode. The gages were mounted on either side of the blade to measure the alternating tensile and compressive stresses and were connected in a bridge circuit, as shown in figure 3. In the investigation of loose mountings, the gap between the blade and the rotor caused by looseness was bridged by a free suspension of insulated stranded wire that negligibly restrained the blade. The rotating portion of the bridge circuit was connected to the stationary instrumentation by means of monel slip rings and silver graphite brushes. A constant direct-current voltage was supplied to the strain-gage bridge and the dynamic output from the strain gages was measured on an electronic voltmeter and a cathode-ray oscilloscope. Frequency of the strain-gage signal was determined by combining it and the output of a calibrated oscillator to form a Lissajous figure. Rotor speed was measured by an electric tachometer.

#### PROCEDURE

The number of blades used in the investigation was limited to three of each the single small and large ball-root-type blades and six of the double ball-root-type blades. Each blade was successively investigated in the tight, loose, and loose-lubricated conditions. In the tight and loose mountings, the blades were inserted dry and as machined; in the loose-lubricated condition, four lubricants were successively used: molybdenum disulfide and grease, graphite and grease, grease alone, and a coating of molybdenum disulfide on the oxidized root surface. The application of grease alone and of combinations of grease with molybdenum disulfide and with graphite was accomplished by forcing the blade into a rotor slot that had been filled with the lubricant. The coating of

molybdenum disulfide on the oxidized surface was prepared by the method reported in reference 1. The blades were oxidized at a temperature of 650° F; while they were at this temperature, a coating of a mixture of corn syrup and molybdenum disulfide was applied to the root. The syrup oxidized, leaving a 0.004- to 0.005-inch coating of carbon and molybdenum disulfide. This layer was carefully scraped until its thickness was such that the blades could be loosely inserted into the rotor slots.

The resonant stress amplitudes in blades of the various root designs and mountings were compared at several orders of rotor speed to determine the relative merits of the several root designs and conditions of mounting. In some instances it was necessary to overspeed the unit to excite resonance at the desired order. A value of the total damping of the blades under investigation was obtained by analyzing the response curves.

#### DISCUSSION AND RESULTS

When a physical member of a system such as a compressor blade is subjected to a periodic disturbance corresponding to a harmonic of its natural frequency, the only restraining force limiting the vibration amplitude is the damping present in the affected system. Damping in compressor blades is produced by four main sources: namely, internal friction of the material; dissipation of vibration energy to the air; transfer of vibration energy through the blade root and rotor to other members in the system having similar natural frequencies (reference 2); and dissipation of vibration energy by the mounting. Authorities have considered material damping insignificant in systems susceptible to high amplitude vibration; aerodynamic damping, being inversely proportional to the density of the blade material, is most effective in light materials and can be detrimental in the case of flutter vibration. In this investigation, material damping and aerodynamic damping were considered to be the same for given test conditions when the various mounting conditions were compared. The vibration-absorption method of providing damping is limited to systems having several components of similar natural frequencies. In compressors it is literally impossible to maintain the blades at desired frequency ratios because of manufacturing tolerances and variations in mounting and in centrifugal force. The uncertainty of vibration-absorption damping effects with a multiblade rotor dictated the use of a single-blade rotor for the present investigation. In a preliminary investigation in which runs with a 34-blade rotor were attempted, large vibration-absorption damping effects were observed. Damping from root mounting can be appreciable even under high centrifugal loading if the blade remains unrestrained.

In the following discussion, for a given condition the vibratory stress as affected by loose mounting and loose mounting with lubricants is compared with the vibratory stress in the same blade when tightly mounted in the rotor. The magnitude of the periodic excitation for a given rotor speed was controlled by the pressure of the air supply to the nozzle. The issuing jet was directed in opposition to the rotation in order to increase the exciting force. A calibration was made of the jet velocity before operation by means of a pitot tube mounted parallel to the nozzle in the free jet stream with the dynamic pressure measured at a point corresponding to the midpoint of the blade surface. The complex flow pattern of the jet on the blade during rotation made difficult an absolute calibration of the exciting force. Because of the flow restriction imposed by the circular housing, it was assumed that the incased air was being rotated at the velocity of the rotor. As a means of analysis, the dynamic pressure  $\rho V^2/2$  is taken as a measure of the exciting force; the quantity is hereinafter called the exciting-force parameter. The factor  $\rho$  is the density of the air in the free jet (lb/cu ft) and  $V$  is the effective velocity (ft/sec), which was calculated by the following equation:

$$V = V_j \cos 45^\circ + V_b$$

where

$V_j$  velocity of jet, (ft/sec)

$V_b$  tip velocity of blade, (ft/sec)

#### Loose Single Ball-Root Mounting

The significance of loosely mounting single ball-root blades in a rotor is shown in figure 4. The data show the effect of the exciting-force parameter on the vibratory-stress amplitude at given orders of excitation of a small and a large ball-root blade. Data are presented for a single blade of each type because data on similar blades agreed within  $\pm 3$  percent of the values shown. The stress amplitudes of the loose blades are compared with those of the tight blades at given orders of excitation used in an attempt to eliminate the harmonic influence. At low speeds (low centrifugal force), some benefit was obtained from looseness at high exciting forces; at low exciting forces, however, the effect was negligible. At the sixth-order excitation (about 9000 rpm), the blades were essentially tight in the mount with only a slight reduction of stress amplitude

at high exciting forces. At the fourth-order excitation (about 15,000 rpm), however, both the loose and tight blades had approximately the same stress amplitude, and at high values of exciting-force parameter they had exceeded the allowable stress of  $\pm 35,000$  pounds per square inch from a fatigue consideration based on combined centrifugal and vibratory stress (reference 3).

Increasing the looseness of the blades in the mount showed no effect on the vibration amplitude. This fact was indicated by fitting a blade to have a free tip movement of 0.004 inch and then increasing the clearance to allow a 0.020-inch tip movement. Data for the 0.004-inch tip movement (not presented herein) agreed with the data for the 0.020-inch tip movement in figure 4 to within  $\pm 1$  percent.

The typical effect of centrifugal force on the first bending natural frequency of blades having tight, loose, and loose-lubricated mountings is shown in figure 5. At low speeds (below 4000 rpm), the loose- and loose-lubricated-blade frequency decreases rapidly because centrifugal force is insufficient to rigidly support the blade. With increased speed, the blade frequencies converge showing increased tightening of the blade.

The use of loose blades has the disadvantage of increased contact stresses in the rotor mount. The loose blades, having relative motion in the mount and being subjected to high centrifugal loading, could, over a period of time, cause a galling condition that would result in fatigue cracks. Figure 6 shows a discoloration in the blade mounting and corresponding blade surface that is characteristic of fretting corrosion. This loose blade was operated at rotor speeds up to 15,000 rpm for a period of about 20 minutes. In the investigation of loose blades reported herein, no serious damage resulted from the high contact stresses.

#### Loose-Lubricated, Single Ball-Root Mounting

Because suppression of blade vibration by means of loose mounting is limited to moderate centrifugal forces, it was believed that this effect could be extended by lubricating the loose blade roots. Also, a proper lubricant would tend to prevent galling effects present in the dry loose mounting. The problem of lubrication involved a method of supply and retention. Molybdenum disulfide has successfully been used (reference 4) as a dry lubricant and was readily adapted to the lubrication of the blade roots.

In this part of the investigation, finely pulverized molybdenum disulfide was mixed with a heavy grease as a means of application.

It is shown in figure 7 that, with equal values of the exciting-force parameter, use of molybdenum disulfide in the loose mounting limited fourth-order vibratory stress to  $\pm 17,500$  pounds per square inch and  $\pm 21,000$  pounds per square inch for the small and large balls, respectively, as compared with approximately  $\pm 38,000$  pounds per square inch for the tight and loose unlubricated blades. In addition to increased damping, at centrifugal loadings normally experienced in current turbojet-engine compressors no fretting corrosion was evident in the lubricated mounting. The increase in root damping due to lubricants is also evident from the fifth-order-resonance curves presented in figure 8. Not only is maximum stress amplitude reduced, but the shape of the curve in the lubricated condition is such as to indicate greater damping. Nonlinear response is evidenced by the asymmetry of the resonance curve.

In an isolated run using only grease as a lubricant, it was found that the vibration-amplitude pattern followed that of the dry loose blades. It was observed after the run that all the grease had been forced out of the mount by centrifugal force and loading. With the molybdenum disulfide and grease, however, a layer of molybdenum disulfide caked on the blades at the contact areas. A mixture of graphite and grease was also used and the results agreed within  $\pm 1$  percent to those of the molybdenum disulfide. The molybdenum disulfide coating on the oxidized surface resulted in a stress amplitude approximately 25 percent higher than that of the molybdenum disulfide and grease. This variation can be attributed to the method involved in applying the molybdenum disulfide coating.

It is obvious that the damping characteristics of loosely mounted blades will vary somewhat with blade-root configuration and loading because of blade size. Increased loading should have less tendency to tighten a loose-lubricated blade using molybdenum disulfide because the coefficient of friction of the molybdenum disulfide decreases with pressure (reference 5). The use of molybdenum disulfide is also favorable at elevated temperatures. Although molybdenum disulfide oxidizes readily at temperatures above  $900^{\circ}\text{F}$ , the friction data of reference 6 indicate that this material produced low friction values as long as an effective subfilm of the lubricant remained.

#### Double Ball-Root Mounting

Even when double ball-root blades are considered to be tight in the mounting and manufacturing tolerances are closely maintained, it is possible that one of the cylindrical sections is loose. The

effect of fourth-order excitation on the vibratory stress of double ball-root blades under various mounting conditions is presented in figure 9. It is noted that a wide variation in the data exists among blades of this type of mounting. The data are for the blade having the highest vibratory stress and the blade having the lowest vibratory stress in the tight- and loose-mounting conditions of the six double ball-root blades investigated. There was no trend as to the high and low stresses between the tight and loose mountings; that is, the blade with the highest stress in the tight condition did not necessarily have the highest stress in the loose condition and the same was true for blades having the lowest stress amplitudes. The vibratory stress amplitudes of the six blades varied but did not approach the maximum allowable vibratory stress. The amplitudes were appreciably lower than those of single ball-root blades for the loose- and tight-mounting conditions at approximately the same rotor speeds and exciting forces. The results appear to indicate that the damping in double ball-root blades was appreciably influenced by the change in geometry of the root with the possibility that the blade had the same upper limit of stress as the single ball-root blades.

In order to investigate further this possibility, the diameter of the lower recess in the rotor was increased so that the blade was retained only by the upper ball. Under this condition, the blade was found to vibrate at the same stress amplitude as that of the single ball-root blades at all orders of vibration and exciting forces.

The effect of lubricating a loose double ball-root mounting is also shown in figure 9. The data presented were obtained with the blade that, when not lubricated, had the highest vibration stress amplitude. For equal values of exciting-force parameter the stress amplitude was reduced from  $\pm 29,300$  pounds per square inch in the dry condition to  $\pm 17,200$  pounds per square inch when lubricated.

#### Damping Measurements

The shape of the resonance curve (fig. 8) is indicative of the damping of a blade as well as of the vibration amplitude. The following table includes average logarithmic values of the decrement of the various single ball-root blades. The maximum and minimum values are included for the double ball-root blade. No damping measurements were taken at the fourth order because of the necessity of prolonged operation at high stress amplitudes. The data have been corrected for the inherent material damping effect by subtracting the material-damping values reported in reference 7. The

resulting values represent the root damping and possible aerodynamic damping; aerodynamic damping is considered constant for comparable conditions. The following equation for damping is based on the frequency difference at 0.707 of the peak of the response curve and was used in the calculation of the values in the table:

$$\delta_{0.707} = \pi \frac{\Delta f}{f_c} 100$$

where

$\delta_{0.707}$  log decrement of damping, percent

$\Delta f$  frequency difference at 0.707 peak amplitude, cycles per second

$f_c$  frequency at peak amplitude of resonance curve, cycles per second

Blade type	Order number	Logarithmic decrement, percent		
		Tight	Loose	Loose-lubricated
Small ball	5	0.9	1.5	9.5
	6	2.2	2.4	8.4
	7	2.2	4.2	9.2
Large ball	5	1.3	1.4	7.5
	6	1.8	2.1	8.1
	7	2.0	3.8	8.3
Double ball	5	11.4-1.9	11.3-2.1	11.7-9.7

#### SUMMARY OF RESULTS

The following results were obtained from an investigation of damping in various blade-root mountings of ball-root-type axial-flow compressor blades:

1. Loose mounting of single ball-root blades offered only a slight reduction of stress amplitude at the sixth-order excitation (approximately 9000 rpm) and at high exciting forces but the effect was negligible at low exciting forces. At the fourth-order excitation, (about 15,000 rpm), however, both the loose and tight blades

had approximately the same stress amplitude and at high values of exciting-force parameter exceeded the maximum allowable vibratory stress of  $\pm 35,000$  pounds per square inch.

2. A change in blade-root clearance that increased free blade-tip movement from 0.004 to 0.020 inch had no appreciable effect on the damping of a single ball-root blade.

3. Loose-blade frequencies were appreciably lower than tight-blade frequencies at low rotor speeds and converged with the frequencies of tight blades at high rotor speeds.

4. Damping in both tight and loose double ball-root blades was influenced appreciably by the variation in geometry of the root.

5. The lubrication of loose-blade mountings with molybdenum disulfide or graphite increased root damping at centrifugal loadings normally experienced in current turbojet-engine compressors and limited the vibratory stress amplitude to a maximum of  $\pm 21,000$  pounds per square inch as compared to approximately  $\pm 38,000$  pounds per square inch for the tight and loose unlubricated blades with equal values of the exciting-force parameter. Because molybdenum disulfide does not oxidize readily at temperatures below  $900^{\circ}$  F, it has satisfactory characteristics at the temperatures encountered in normal compressor operation.

6. Loose blade mounting showed evidence of fretting corrosion at contact surfaces under high centrifugal loading; however, in the loose-lubricated condition, no evidence of fretting corrosion was observed.

Lewis Flight Propulsion Laboratory,  
National Advisory Committee for Aeronautics,  
Cleveland, Ohio.

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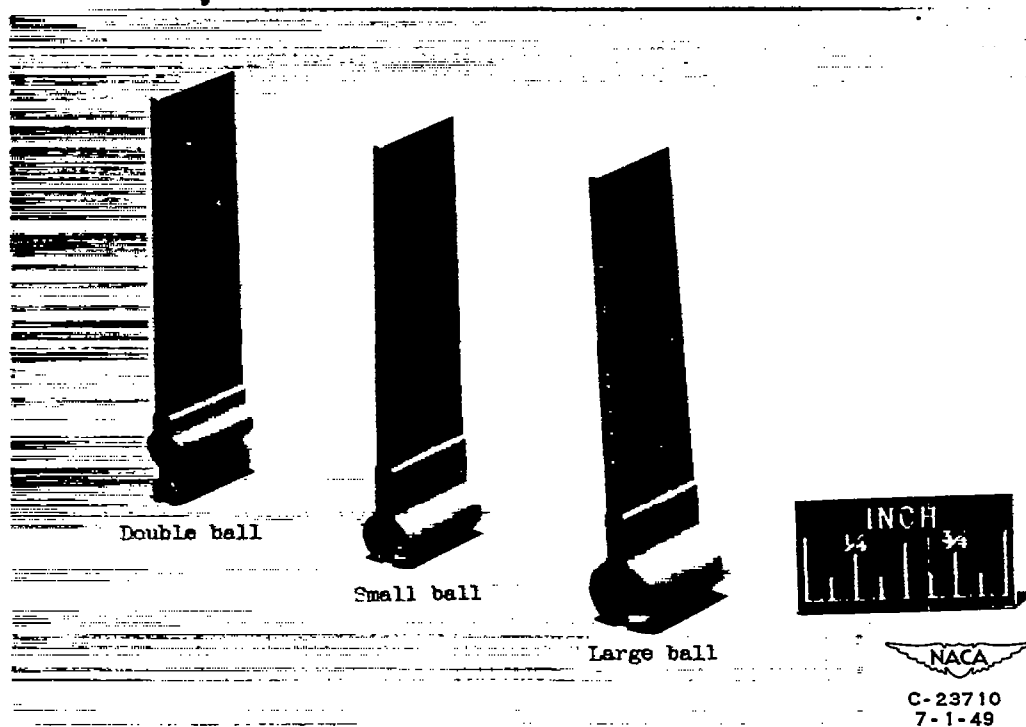


Figure 1. - Ball-root configurations investigated.



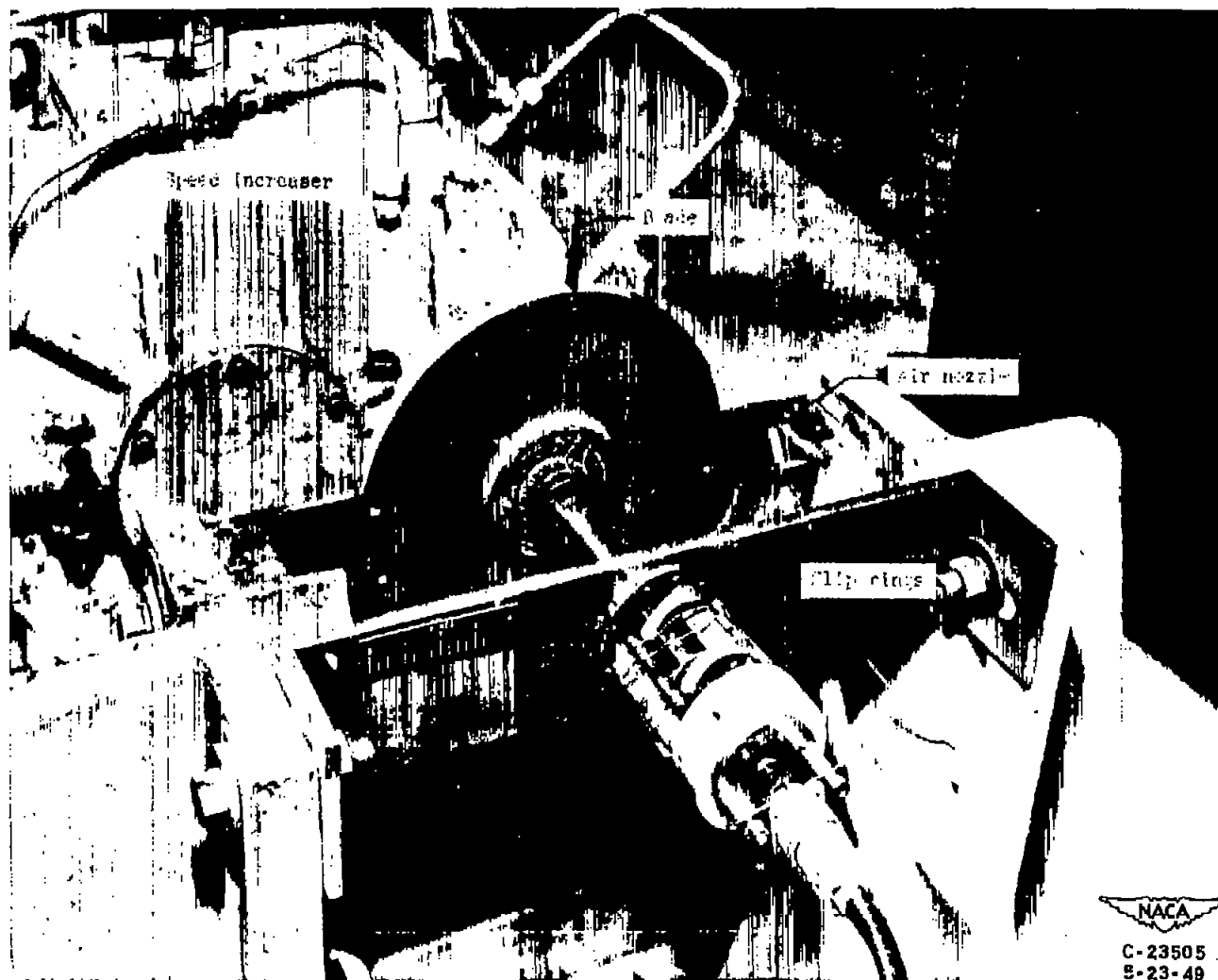


Figure 2. - Blade-vibration spin rig with upper housing removed.



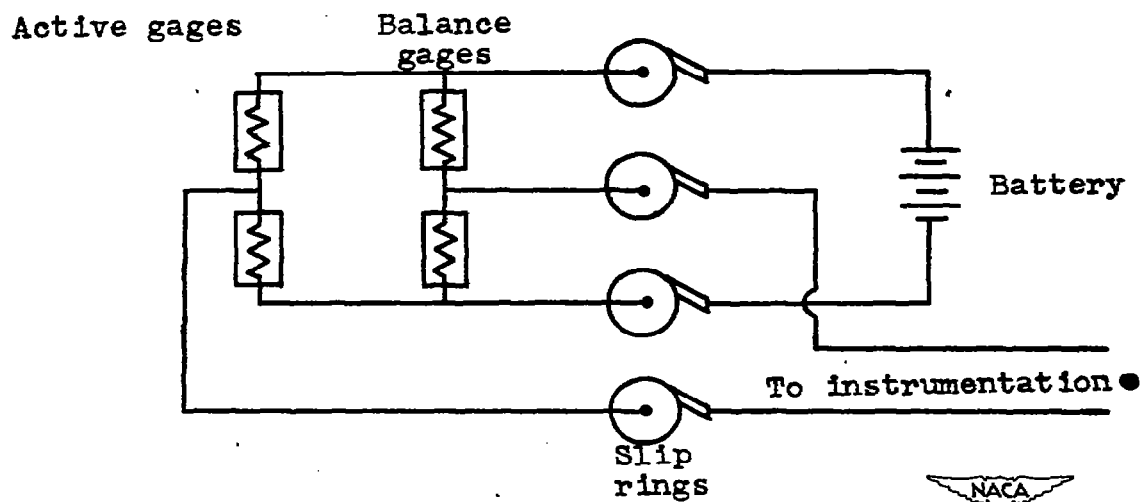
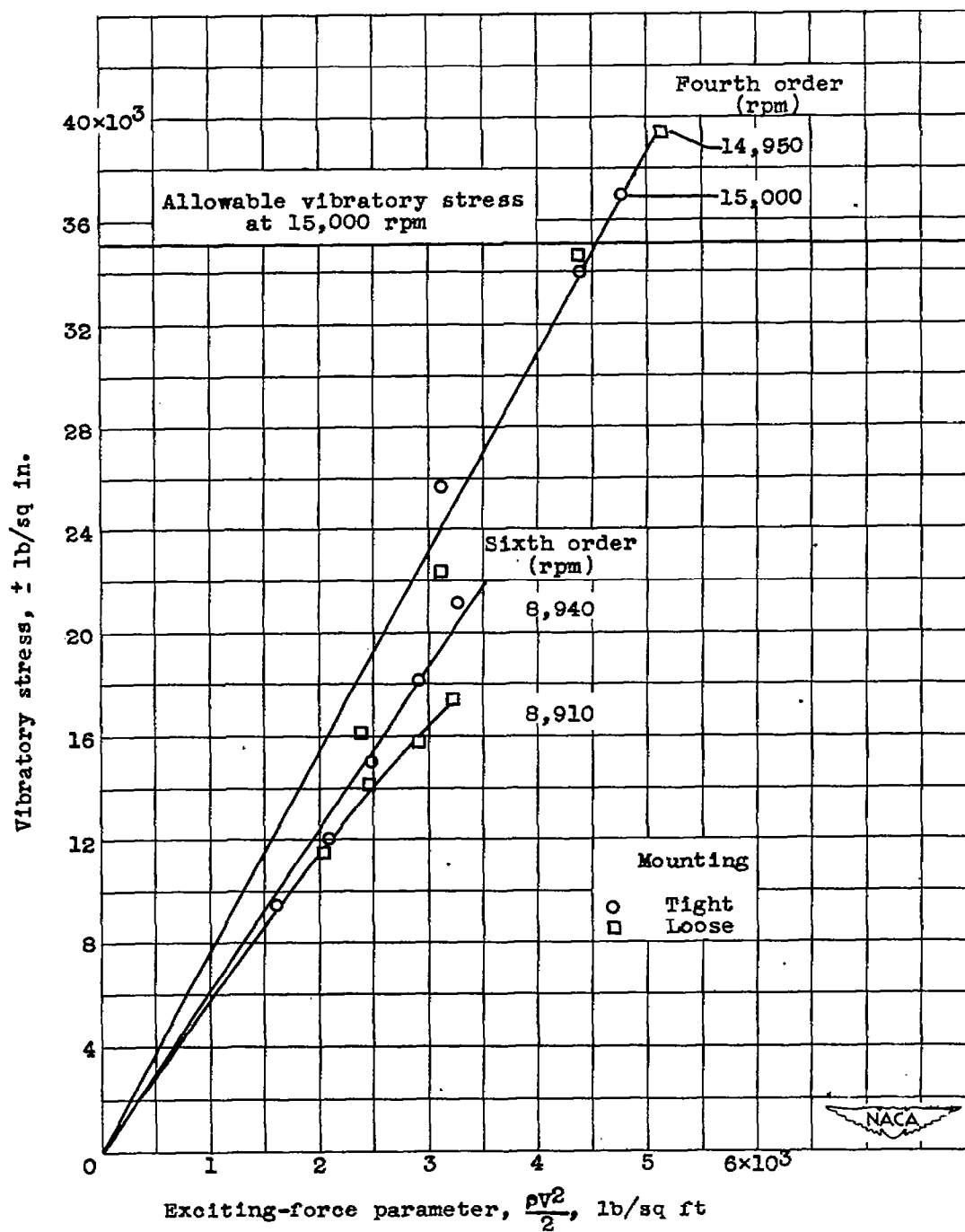
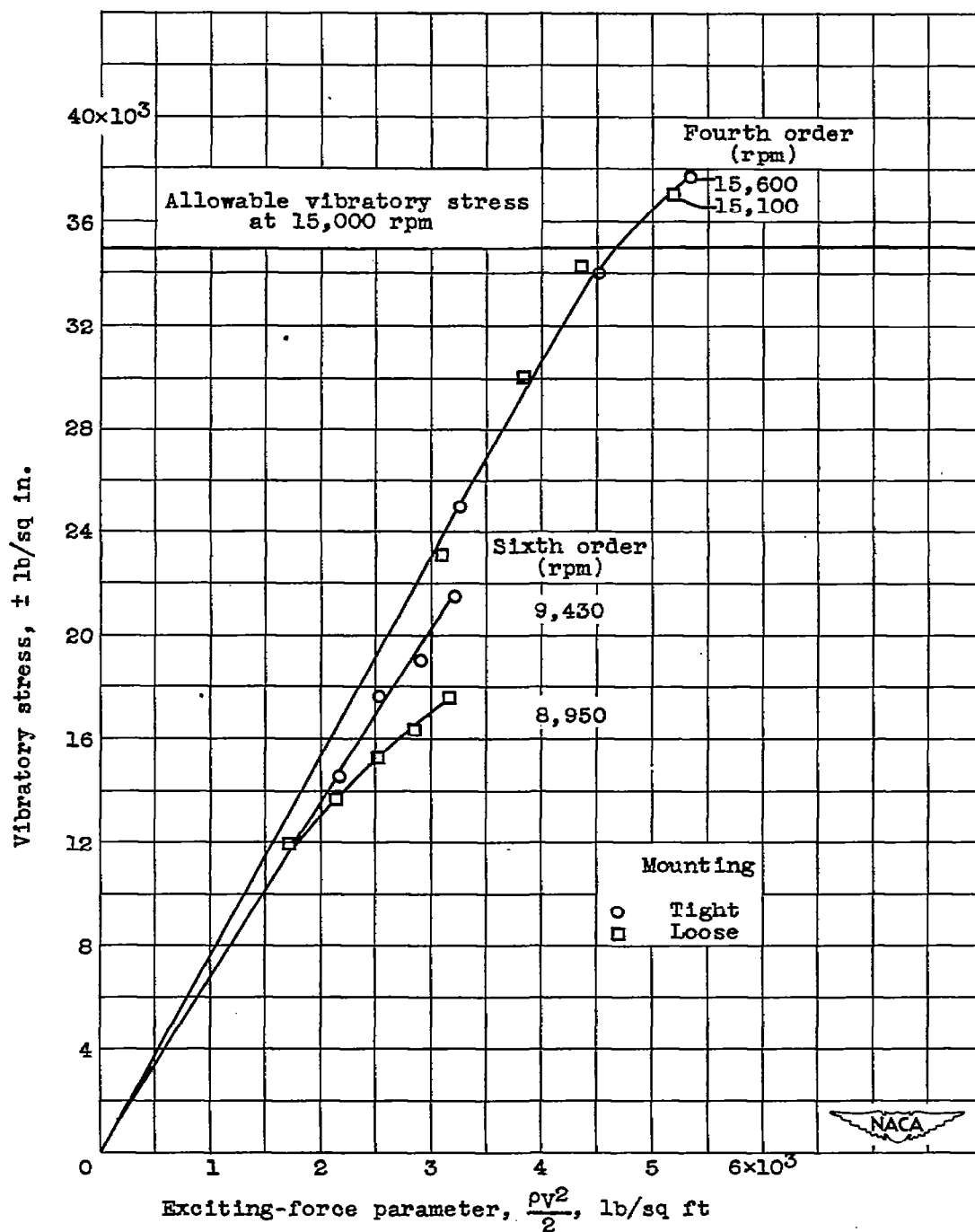


Figure 3. - Schematic diagram of strain-gage bridge circuit.



(a) Small ball.

Figure 4. - Effect of exciting force on vibratory stress for loose and tight blade mounting under centrifugal loading.



(b) Large ball.

Figure 4. - Concluded. Effect of exciting force on vibratory stress for loose and tight blade mounting under centrifugal loading.

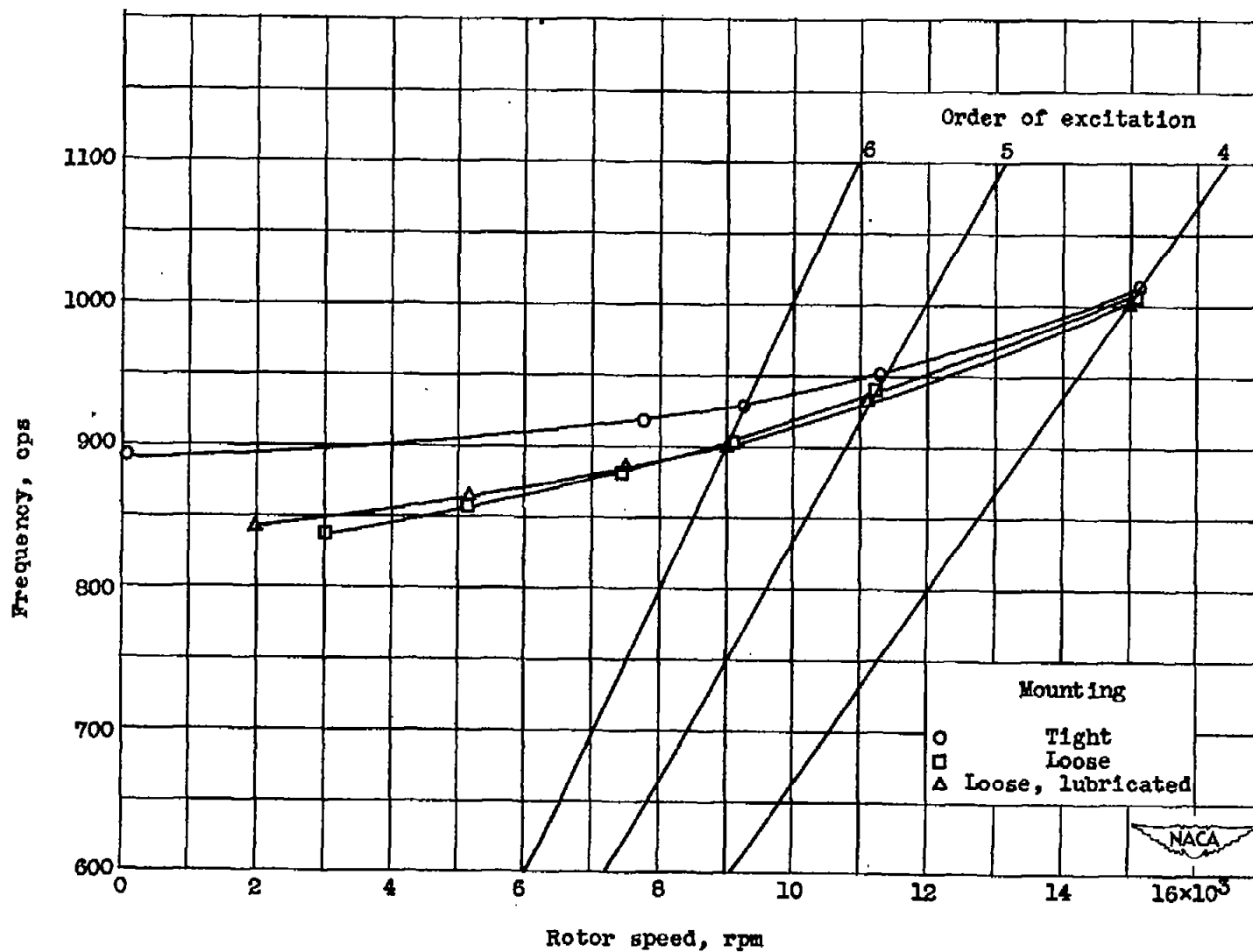
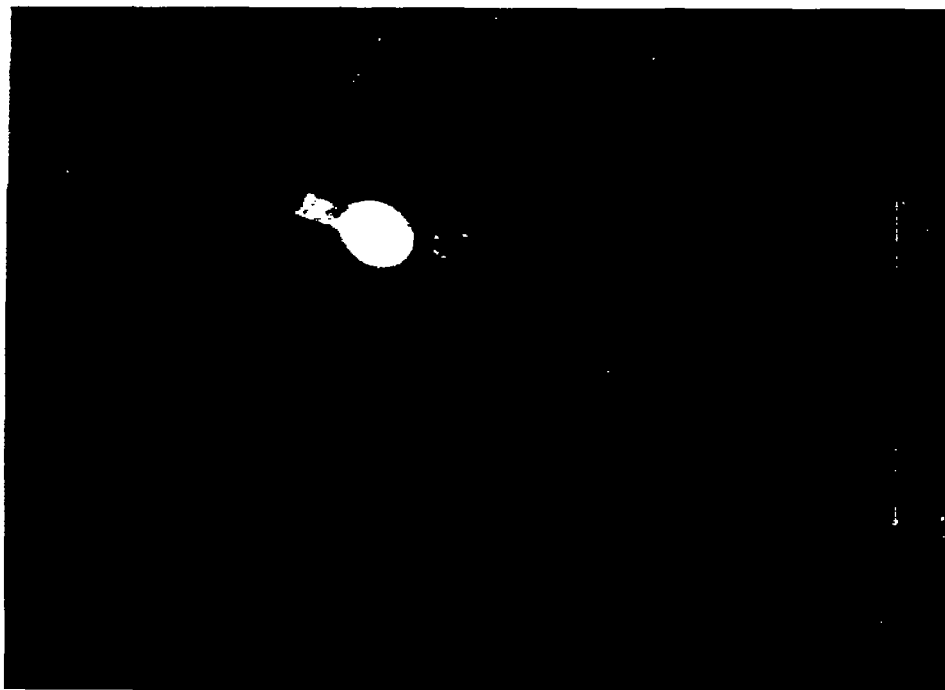
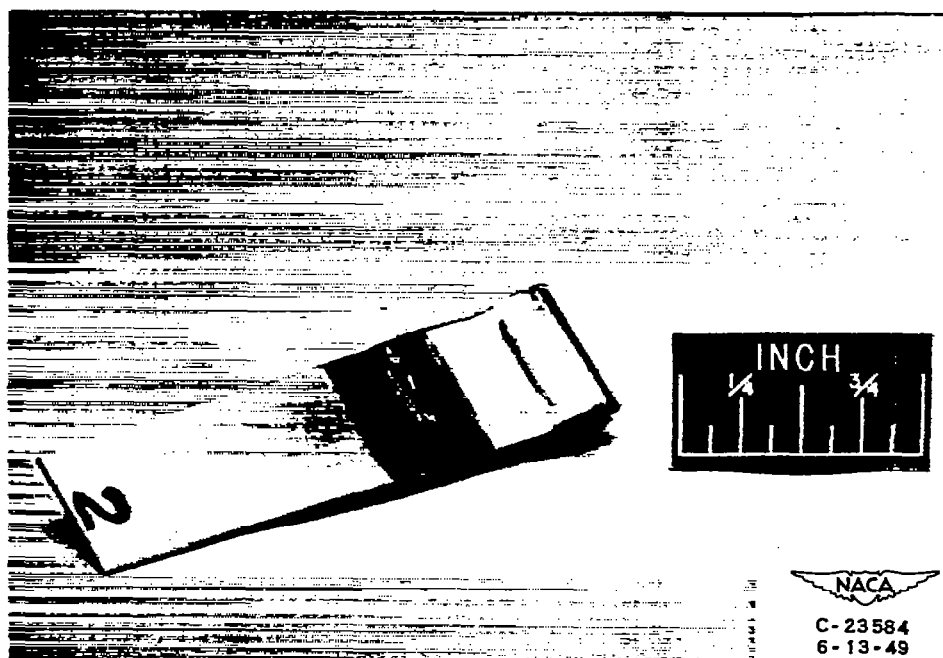


Figure 5. - Effect of centrifugal force on first bending natural frequency of small ball-root blade mounted with varied test conditions.

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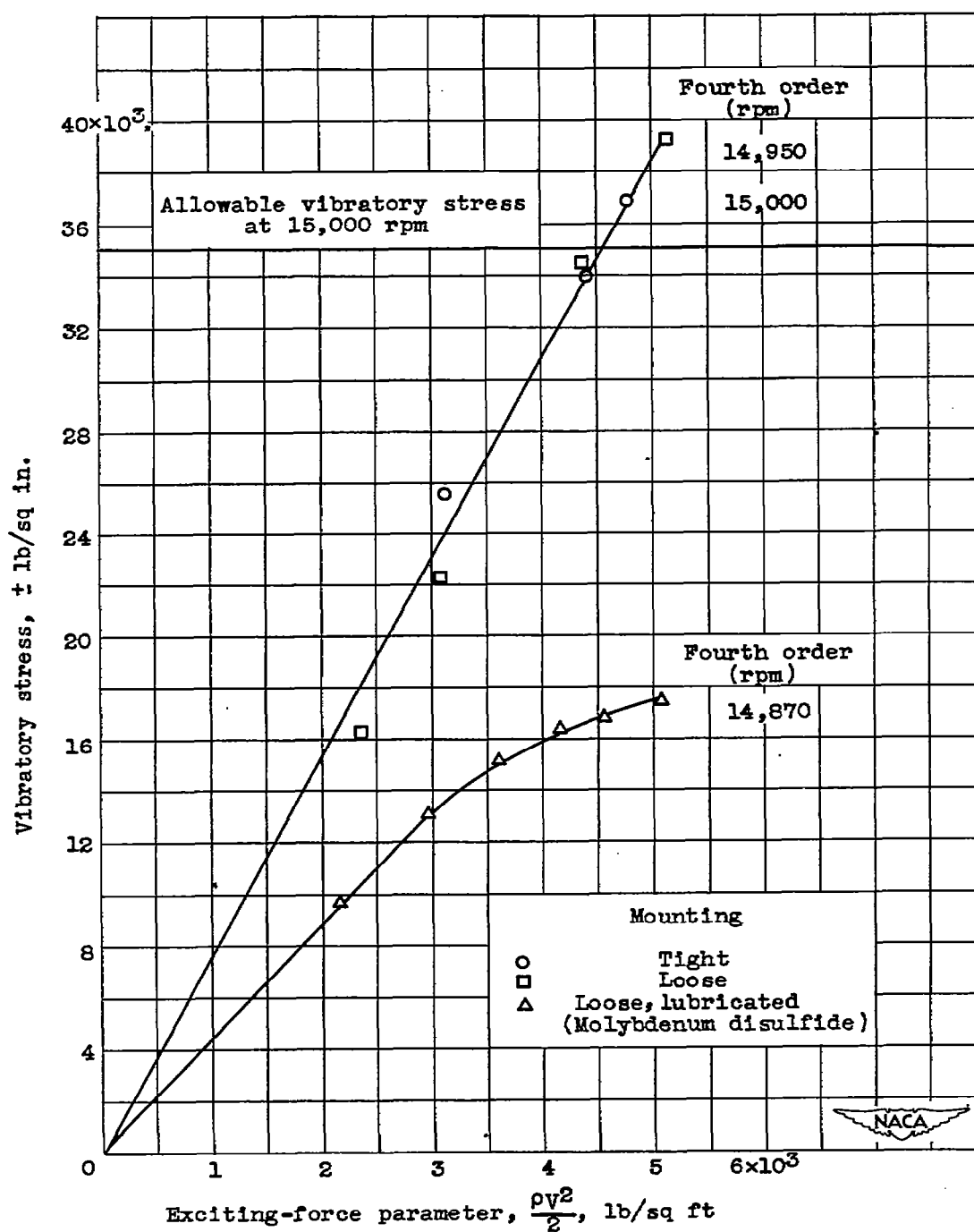
(a) Rotor



(b) Blade

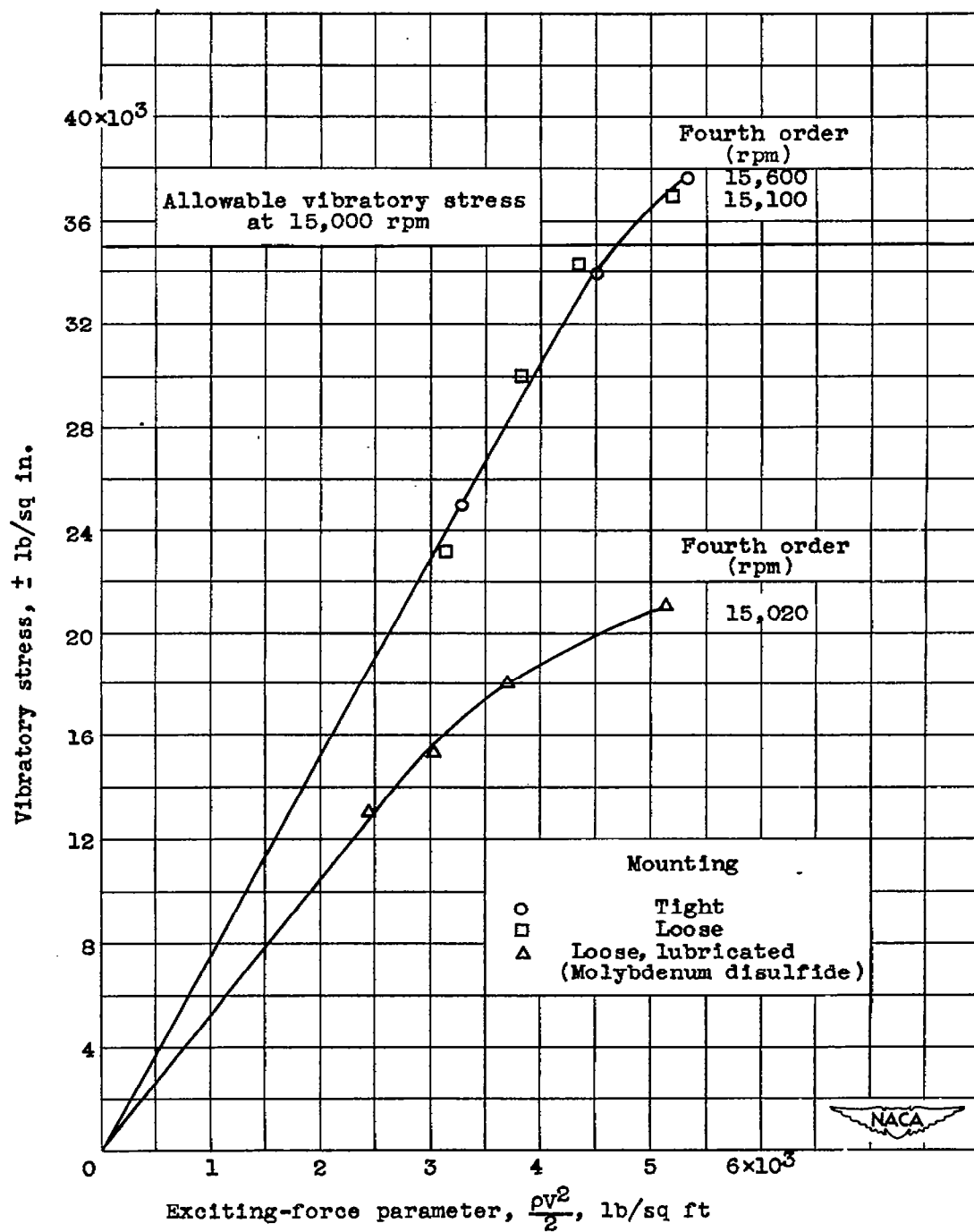
Figure 6. - Fretting corrosion caused by dry loose mounting.





(a) Small ball.

Figure 7. - Effect of exciting force on vibratory stress for loose-lubricated blade mounting under centrifugal loading.



(b) Large ball.

Figure 7. - Concluded. Effect of exciting force on vibratory stress for loose-lubricated blade mounting under centrifugal loading.

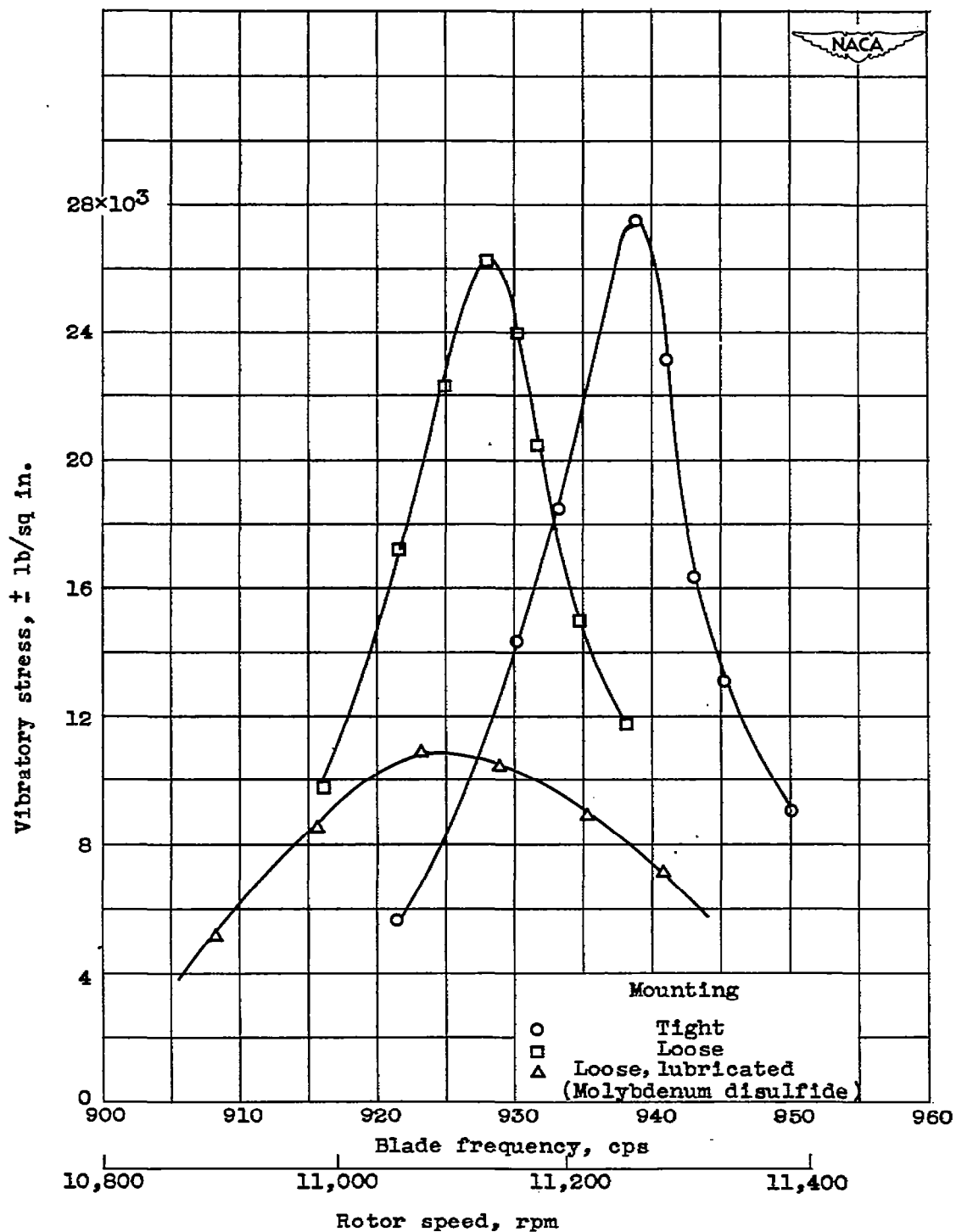


Figure 8. - Typical fifth-order-resonance curves showing effect of mounting single small ball-root blade on stress amplitude.

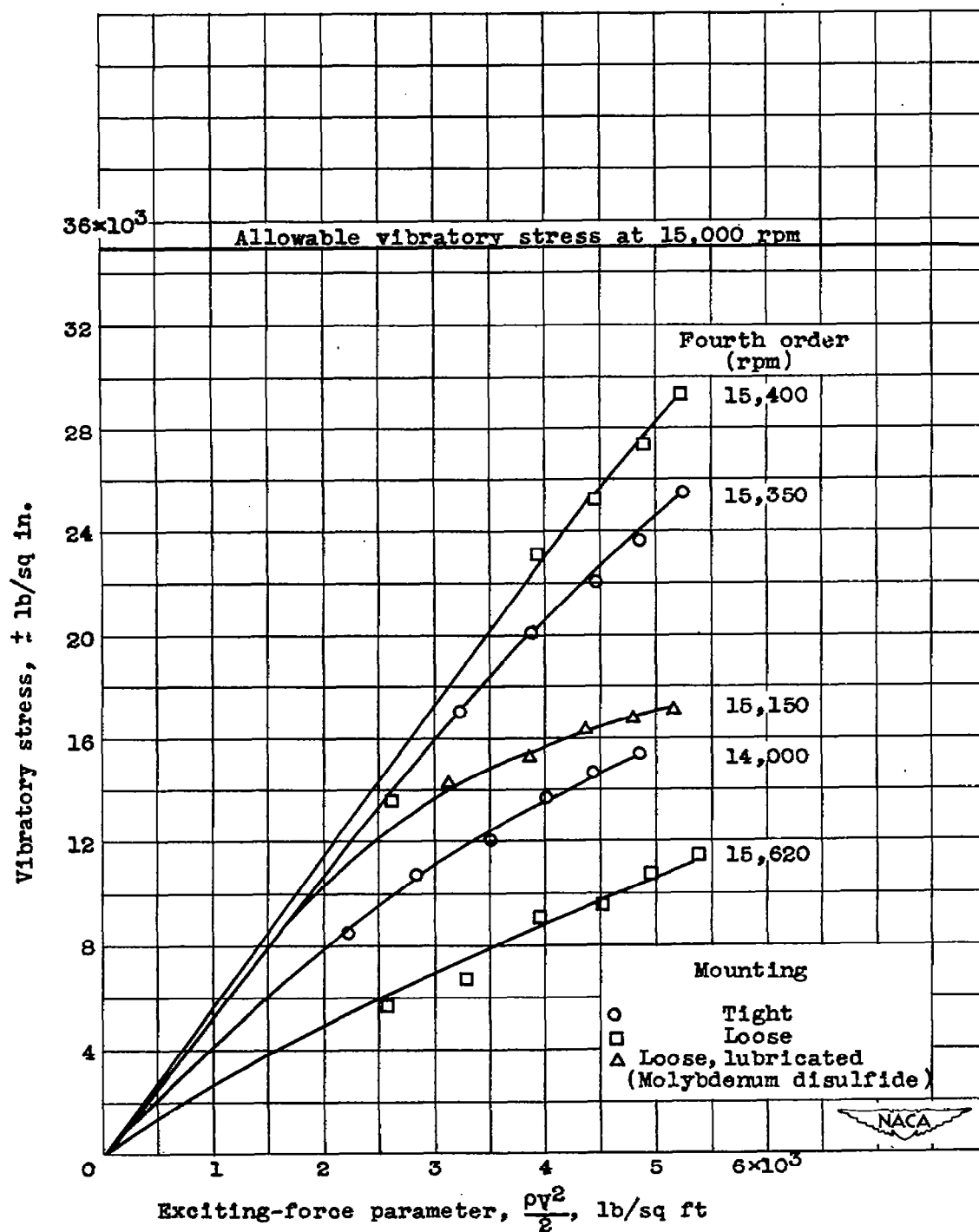


Figure 9. - Effect of exciting force on vibratory stress for double-ball-root blades under centrifugal loading showing highest and lowest stress amplitudes of six blades investigated.